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A COMPARISON OF ANALYTICALLY AND EXPERIMENTALLY

DETERMINED ISOTHERMAL PRESSURE LOSSES IN A

HEAT-EXCHANGER INSTALLATION

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the calculated component pressure losses will yield the over-
all pressure loss in a system where the magnitude and loca-
tion of interference effects are largely unknown. Interfer-
ence effects are defined as those occurring where disturbances
in one component cause the flow distribution at the entry of
another component to be unsymmetrical, thereby affecting the
pressure loss.

This report contains an analysis in which the material
in the bibliography of reference 1 was used to calculate the
pressure losses in a heat-exchanger installation. The iso-
thermal pressure losses in the installation were measured for
the purpose of providing an experimental verification of the
analytical predictions. The comparison indicates the validity
of the analysis and the magnitude of the interference effects
in the test installation.

DESCRIPTION OF THE HEAT-EXCHANGER INSTALLATION

Heat exchangers of the type reported herein have found
particular application in the field of ice prevention and
cabin heating where they are used to supply an adequate and
reliable source of heated air by utilizing the waste heat in
the engine exhaust gas. The heat-exchanger installation here-
in described was developed for use in the ice-prevention sys-
tem of a large airplane. The heat exchanger is located in the
exhaust-gas ducting between the engine and two turbosuper-
chargers which are arranged for single or parallel operation.
Turbosupercharger 1 is blocked off by a butterfly valve when
turbosupercharger 2 is operating alone. When both are operat-
ing in parallel, the flow of exhaust gas is adjusted so as to
be divided equally between the turbosuperchargers.

General views and construction details of the heat ex-
changer are shown in figures 1 and 2, respectively. In a
cross-flow plate-type heat exchanger, such as the one tested,
the exhaust-gas and air flow at right angles in alternate pas-
sages between formed steel plates, each plate thus forming an
interface through which heat is transferred from the exhaust
gas to the air. At design conditions of 155 miles per hour
and 18,000 feet pressure altitude, a total of 800,000 Btu per
hour is to be transferred from 12,000 pounds per hour of ex-
haust gas to an equal amount of air.

The headers shown in figures 3 and 4 serve as transition
pieces in ducting air and gas into and out of the heat

exchanger. The branches in the exhaust-gas outlet header
lead to the two turbosuperchargers.

ANALYSIS

The isothermal pressure losses in each component of the
test installation were calculated as follows: The shape of
the component was compared to those on which data (usually
shown in graphs) were published, and where similarity existed
the data were applied to the component. The loss factor cor-
responding to the dimensions of the component was then read
from the reference data and multiplied by the local velocity
head to obtain the pressure loss. Where losses in a compo-
nent resulted from a combination of effects (i.e., diffusion,
turning, contraction) each effect was isolated, the corre-
sponding loss calculated, and the results added to give the
total loss.

Air-Side Losses

Inlet-header losses.- The air inlet header is shown in
figure 3(a) to consist of a vaned elbow superimposed on a
diffuser. The loss in the elbow was computed using data on
vaned elbows published in reference 2 where the pressure-loss
factor is plotted as a function of the s/c or gap/chord
ratio in the vanes. This value ranged from 0.48 to 0.62, and
the corresponding loss is approximately 25 percent of the ve-
locity head at the header entrance. At the design flow rate
of 12,000 pounds per hour, the velocity head is 0.90 inch of
water and the turning loss is 0.225 inch of water.

The loss caused by the expansion in area was computed
using data on straight diffusers which appear in reference 3.
If the expansion is assumed to occur principally in the vanes,
an effective angle of divergence, based on the length of the
vanes (3 to 7 in.) and the ratio between the inlet and outlet
areas ($A_2/A_1 = 2.3$), has a value between 11° and 23°. The
corresponding loss factor is about 20 percent of the change
in velocity head through the vanes, which at the design flow
rate is 0.73 inch of water. The expansion loss is then 0.145
inch of water.

The total air-inlet-header loss is then the sum of the
turning and diffuser losses - or 0.37 inch of water.

Heat-exchanger losses.- The loss in the heat exchanger
was assumed to result from friction on the plate surfaces and
expansions in the three corrugations running transverse to
the air flow and in the outlet end of the plates. (See fig.
2.) The friction loss was calculated using the Fanning equa-
tion (reference 4) and is 1.12 inches of water at the design
flow rate.

Since the expansions have an included angle larger than
50°, the losses are approximately equal to those in sudden
expansions. A formula for such losses appears in reference 3
and in this case the total loss for all four expansions is
2.02 inches of water.

The total heat-exchanger loss is 3.14 inches of water at
the design flow rate.

Outlet-header losses.- Since the contraction is gradual,
the principal loss occurring in the outlet header was assumed
to result from the elbow. Data in references 2 and 5 were
used. For a value of s/c of about 0.3, the loss factor is
0.2. As a check, if the vanes are omitted, the loss corre-
sponding to a mean-elbow radius/diameter ratio R/D of 1 has
a value of 0.25. This is considered to be good agreement and
the loss corresponding to a velocity head of 0.85 inch of
water is about 0.17 inch of water.

Over-all loss.- By adding together the above calculated
values a result of 3.68 inches of water is obtained for the
pressure loss in the air side of the installation at design
flow rate.

The calculated values of pressure losses in the air-
side ducting are plotted in figure 5.

Gas-Side Losses

Inlet-header losses.- As the air (during isothermal
tests) enters the inlet header (fig. 4(a)), it is immediately
turned through approximately 90° in an elbow having a circu-
lar cross section. Following the turn it passes through an
expansion in duct area. An elbow which is followed by a
straight section of duct causes a lower pressure loss than
one which is not, because of recovery in the duct. This ef-
fect is shown in reference 5 for rectangular ducts. The same
reference presents data only on circular elbows which are
followed by straight ducts; therefore, in order to apply

these data to the gas inlet header, the loss was increased
by the comparable difference between the curves for rectangu-
lar ducts.

Thus, for an R/D of 0.85, the elbow loss factor is
0.33, and by adding 50 percent (from comparison with rectan-
gular ducts) the loss becomes 50 percent of the velocity head -
or 0.52 inch of water.

Similar to the above calculation, the effect of the el-
bow on the diffuser loss is expected to be appreciable in
this case. Because of the separation of flow at the inner
radius of the elbow, the air is not diffused uniformly, and
it is expected that the use of data on straight diffusers
will result in an underestimate of the loss. In this case
the loss was assumed to be equal to that in a sudden expan-
sion. The loss was calculated to have a value of 40 percent
of the change in velocity head, or about 0.35 inch of water.

The total calculated loss in the inlet header is then
the sum of the two losses, or 0.87 inch of water at the de-
sign flow rate of 12,000 pounds of air per hour.

Heat-exchanger losses.- In effect, there are 88 parallel
passages through the gas side of the heat exchanger, since
the 22 passages between the plates are each divided into four
by the corrugations shown in figure 2. It is assumed that
the flow is equally divided between the passages, and that
the calculation of the pressure drop through any one passage
is the same as that of the entire exchanger.

Using the Fanning equation, as for the air side, the
friction pressure drop was calculated to be 0.56 inch of
water at the total flow rate of 12,000 pounds of air per hour
through the heat exchanger.

The only expansion loss occurs at the outlet end of the
passage and, as before, its loss is considered to be equal
to that of a sudden expansion. Since the area ratio is 2.3,
this loss amounts to 0.28 inch of water.

The sum of the heat-exchanger losses is then 0.84 inch
of water at the design flow rate.

Outlet-header losses.- As shown in figure 4(b), the gas
outlet header consists of a combination wye and a contraction.
This shape is too irregular to lend itself to elementary anal-
ysis. It will be noted, however, that a 90° turn occurs in

the branch to turbosupercharger 1. If, under the condition
of operation when the flow is equally divided between the two
turbosuperchargers, the total pressures are the same in each
branch upstream of the elbow, then any difference downstream
should be the result of the turning loss in the elbow. This
is the only loss that was calculated for the gas outlet header.

A flow rate of 6,000 pounds per hour through the elbow
corresponds to a total flow rate of 12,000 pounds per hour
through the heat exchanger, and for an R/D of 0.75 in the
elbow the loss is 35 percent of the velocity head - or 0.32
inch of water.

Over-all losses.- The over-all pressure loss on the gas
side has been calculated only for the condition where the flow
is equally divided between the ducts leading to the two turbo-
superchargers. This value for the branch of the system lead-
ing to turbosupercharger 2 is the sum of the inlet-header and
heat-exchanger losses, and for the branch leading to turbo-
supercharger 1 is the sum of the inlet-header, heat-exchanger,
and outlet-elbow losses. The two values are 1.71 and 2.05
inches of water, respectively, at the design flow rate.

The calculated values of pressure losses in the gas-side
ducting are plotted in figure 6.

DESCRIPTION OF TEST APPARATUS

The heat exchanger and headers were assembled and tested
for isothermal pressure losses in the arrangements shown in
figures 7 to 10. As seen in these figures, a number of dif-
ferent arrangements were tested. It was possible in this
manner to determine not only the pressure losses in the vari-
ous parts, but also the magnitude of the interference effects.

The butterfly valve upstream of the venturi meter in the
duct to turbosupercharger 1 (fig. 10) was shown by the tests
to have a negligible effect on the accuracy of the venturi
under the conditions of the tests.

The pressure rakes and traversing shielded total- and
static-pressure tubes shown in figure 11 were used in making
the pressure measurements. The manometers on which these
pressures were indicated are shown in figure 8. In all cases,
the multiple-tube manometers were used with the pressure rakes
and micromanometers with the traversing tubes shown in figure

11(b). These micromanometers are mechanically driven and are set up to indicate a single differential pressure to a least count of 1 millimeter of fluid (in this case, water). The use of multiple-tube manometers, together with the pressure rakes, made it possible to obtain instantaneous flow patterns on the entire diameter of the ducts. The shielded total pressure tubes were used in regions where the direction of flow was uncertain. These tubes will indicate the correct total pressure when yawed through angles up to 60° from the direction of flow.

TEST PROCEDURE

The measured values of pressure losses in the heat-exchanger installation were obtained from the following arrangements. The general purpose of the procedure was to obtain as many cross checks on the data as were practical.

(a) The arrangements shown in figure 8 were used in obtaining the data on over-all pressure losses. It was in this form that the installation was originally designed.

(b) The losses in the heat exchanger were determined with the arrangements shown in figure 7, the straight headers allowing measurement of the losses as unaffected by the headers.

(c) Figures 9(a) and 10(a) show the arrangements in which the header losses were measured with the heat exchanger in place. These arrangements are identical with those in figures 8(a) and 8(b) except that traversing sections have been inserted between the headers and the heat exchanger.

(d) Figures 9(b) and 10(b) show arrangements used in measuring header losses with the heat exchanger removed.

The above-outlined procedure has made it possible to use the data from two independent tests in plotting the values of pressure loss in each part. In the case of the exhaust-gas side of the installation, two conditions of operation were investigated. Referring to figure 10, under one condition the branch to turbosupercharger 1 was blocked off and all air was directed to turbosupercharger 2, and, under the second

condition, the air flow was equally divided between the two turbosuperchargers. The over-all pressure losses were measured under both conditions. The radial locations at which pressure measurements were obtained in the round ducts are given by the rake details in figure 11(a). For the rectangular ducts in which the traverses were made, the duct cross section was assumed to be divided into 25 equal-area squares and pressure measurements were taken at the center of each square.

TEST RESULTS

The experimentally determined values of pressure losses in the air side of the installation are presented in figure 5 and those in the gas side in figures 6(a) and 6(b). The difference between the pressure losses determined with arrangements (c) and (d) in the test procedure was negligible and test points from both arrangements are presented in the curves in these figures.

Each test point represents the difference between average total pressures at stations upstream and downstream of the duct component. The average pressure at a station was obtained from 20 total-pressure readings in the case of the round ducts (for locations, see fig. 11(a)) or from 25 total-pressure readings at the centers of as many equal-area rectangles in the rectangular ducting.

In order to show the effect of the headers on the pressure loss in the air and gas passages of the heat exchanger, additional data were taken using the arrangements shown in figures 9(a) and 10(a). These data are compared in figure 12 with those taken using the arrangement in figures 7(a) and 7(b). Likewise, to show the effect of the traversing sections (which were installed between the headers and heat exchanger to measure the header losses) on the over-all pressure loss, data taken from the arrangements shown in figures 8(a), 8(b), 9(a), and 10(a) are plotted in figure 13.

ACCURACY OF MEASUREMENTS

From a consideration of the readability of the manometers, accuracy of venturi constants, and flow conditions in the installation, the flow rates are thought to be accurate to within ±2 percent, and the values of over-all pressure losses within about ±5 percent of the measured values.

Because of large-scale turbulence coupled with complete reversals of flow in the regions between the headers and heat exchanger, the accuracy of pressure-loss measurements in the headers is believed to be of the order of ±10 percent of the measured values.

DISCUSSION

A comparison of the analytical and experimental results in figures 5, 6(a), and 6(b) indicates that the predicted values of the losses were, in general, slightly low. The agreement being generally closer than 20 percent indicates that the analysis is applicable and is evidence of the fact that the interference effects did not cause a large increase in the losses in this installation over the predicted values in which the interference was assumed negligible.

The experimental and analytical results from figures 5, 6(a), and 6(b) are presented in bar-graph form in figures 14 and 15 for comparison of the losses in the component parts. The agreement between the experimental and analytical values for the over-all pressure loss is better than those for the component parts because of the more accurate measurement of over-all pressures.

When the values of measured pressure losses in the component parts of the gas side of the installation are added together, the summation exceeds the measured value of over-all pressure loss by the amount shown in figure 15 as experimental error in measuring component losses. This difference is attributed to the difficulty of making accurate measurements in the regions between the gas headers and heat exchanger. A similar difference between the summation of individual and over-all losses is not evident in the air ducting (fig. 14) where the large separation effects are not present and resulting measurements are more accurate.

The difficulty in obtaining accurate measurements in the region between the headers and heat exchanger is further emphasized by the comparison of gas-side data for the two conditions shown in figure 12 where the installation headers appear to reduce the pressure loss in the gas side of the heat exchanger. Such an effect is improbable and this difference is attributed to inaccuracies in measurement.

A header of the type shown in figure 4(a) is undesirable, not only because of the excessive pressure loss (in fig.

15 the pressure loss in the inlet header is shown to be larger
than that in the heat exchanger), but also because of the un-
even distribution of flow into the heat exchanger. Flow dis-
tribution in the various parts of the installation is shown
in figures 16 to 19. It is to be noted that, while the heat
exchanger has an appreciable damping effect on the distribu-
tion, figure 18 shows the uneven distribution caused by sep-
aration in the inlet header to persist into the region down-
stream of the heat exchanger.

One of the values in this type of analysis is in the lo-
cation of losses which cannot be isolated experimentally.
The test results can only indicate the magnitude of the loss
through the duct component, and offer no clue to the cause of
the loss or the distribution of the loss among several causes.
Thus, without an analytical approach, the large contribution
(fig. 14) of the expansion losses in the exchanger to the
over-all loss across the exchanger might not be suspected.
The corrugation expansion losses in the air side, which are
not present in the gas side, plus the increased friction re-
sulting from narrower gaps on the air side combine to cause
the air-side loss to be over five times as large as the gas-
side loss.

The agreement between the analytical and experimental
values of over-all pressure loss on both sides of the instal-
lation shows that it is possible to predict analytically the
magnitude of the losses in a ducting system of this type.
However, it is obvious that there is a need for investiga-
tions of interference effects, particularly in expanding el-
bows of the type appearing in this installation.

In the ducting arrangements used in the present inves-
tigation, the air was drawn into the installation through
short bell-entry ducts. Air delivered to the test region
through such entries has a low-turbulence level and there is
little opportunity for the growth of boundary layers at the
walls. Several investigators have shown that when the flow
is more disturbed and when thick boundary layers are present
in the supply duct, the losses, particularly in diffusers,
are increased because of the more favorable conditions for
the occurrence of separation. It is probable that the inlet-
header losses in this installation would increase under such
conditions; however, it is not possible at the present time
to predict their magnitudes. This points to the need for
systematic investigations of the effects of initial turbulence
and flow conditions.

CONCLUSIONS

1. Isothermal pressure losses can be predicted analytically in the type of ducting system which may be broken down into the elementary forms of elbows, diffusers, and straight ducts for which pressure-loss data are published.

2. For duct installations similar to that tested, in which the interference effects may be reasonably assumed to be small or negligible, the summation of the calculated losses for the individual components will be in close agreement with experimentally determined values of the over-all pressure loss.

3. The large-scale separation occurring in poorly designed expanding elbows results in uneven flow distribution and increased pressure loss. The results of this investigation indicate that the losses in such elbows can be predicted analytically with a fair degree of accuracy, but experimental values of the losses are subject to the uncertainties of pressure measurements in regions of uneven flow distribution.

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      National Advisory Committee for Aeronautics,
            Moffett Field, Calif.

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AAL
4812

(a) Air passages

AAL
4813
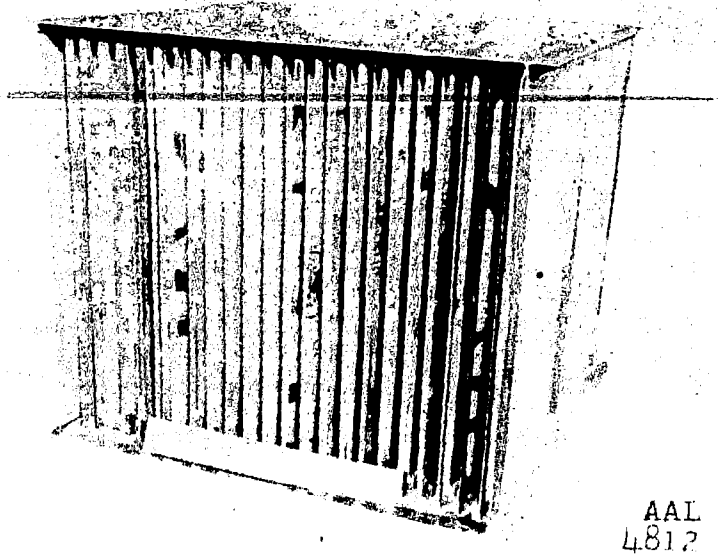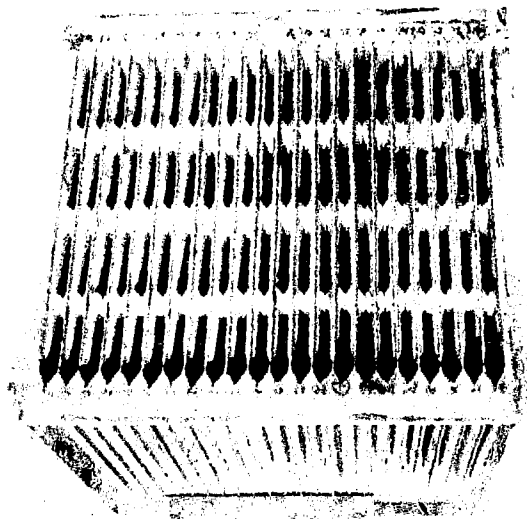
(b) Gas passages

Figure 1.- The exhaust-gas-to-air heat exchanger.

FIGURE 2.- DETAILS OF THE HEAT EXCHANGER

PLAN

NATIONAL ADVISORY COMMITTEE
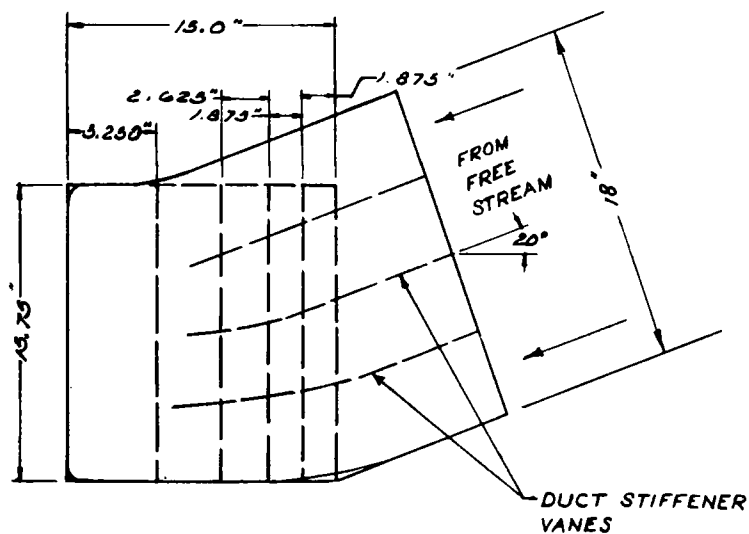FOR AERONAUTICS

NOTES:
1. MATERIAL: CORROSION & HEAT RESISTING
   STEEL (AN QQ ST5T ISS)
   HEATER PLATES. - .032" THICK
   HEATER FLANGES - .051" THICK
2. WEIGHT OF HEATER - 96 LBS.
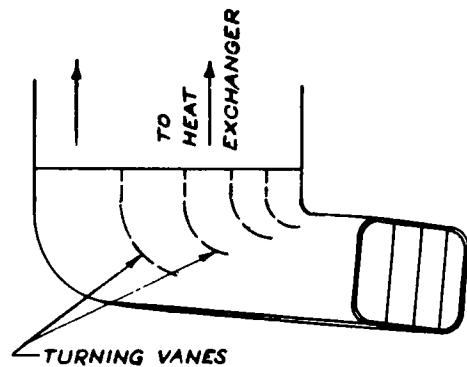
SECTION "A-A"

SECTION "C-C"

SECTION "B-B"

END VIEW OF
AIR PASSAGES

END VIEW OF
GAS PASSAGES

23 AIR PASSAGES
23 EQ. SPA.= 15.745

22 EXHAUST GAS PASSAGES
22 EQ. SPA.= 15.061

Fig. 2

TO HEAT EXCHANGER

TURNING VANES

NOTE: ARROWS INDICATE DIRECTION OF FLOW

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15.0"

2.625"

1.875"

3.250"

1.875"

FROM FREE STREAM

18"

20°

15.75"

3.62"

DUCT STIFFENER VANES
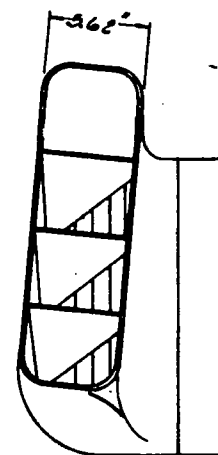
(a) INLET HEADER

FIGURE 3.- HEADERS WHICH DUCT AIR INTO
AND OUT OF THE HEAT EXCHANGER

FROM HEAT EXCHANGER

15.0"

PLUG

TURNING VANES

12.0"

TO WING ICE PREVENTION SYSTEM

NOTE: ARROWS INDICATE DIRECTION OF FLOW

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15.75"
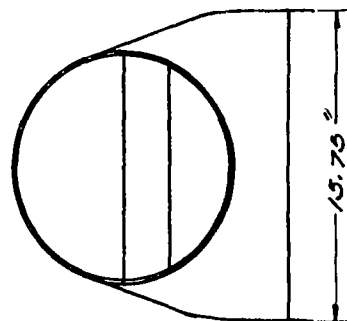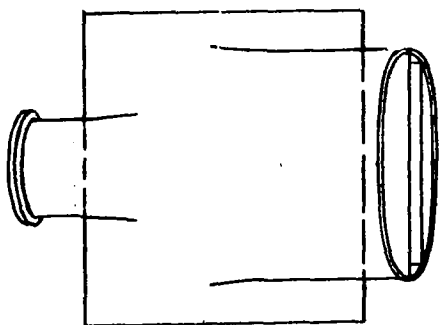
(b) OUTLET HEADER

FIGURE 3.- CONCLUDED

TO HEAT EXCHANGER

15"

NOTE: ARROWS INDICATE DIRECTION OF FLOW

$8\frac{1}{2}"$ R (APPROX.)

$10\frac{3}{4}"$ DIA.

FROM ENGINE

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15"
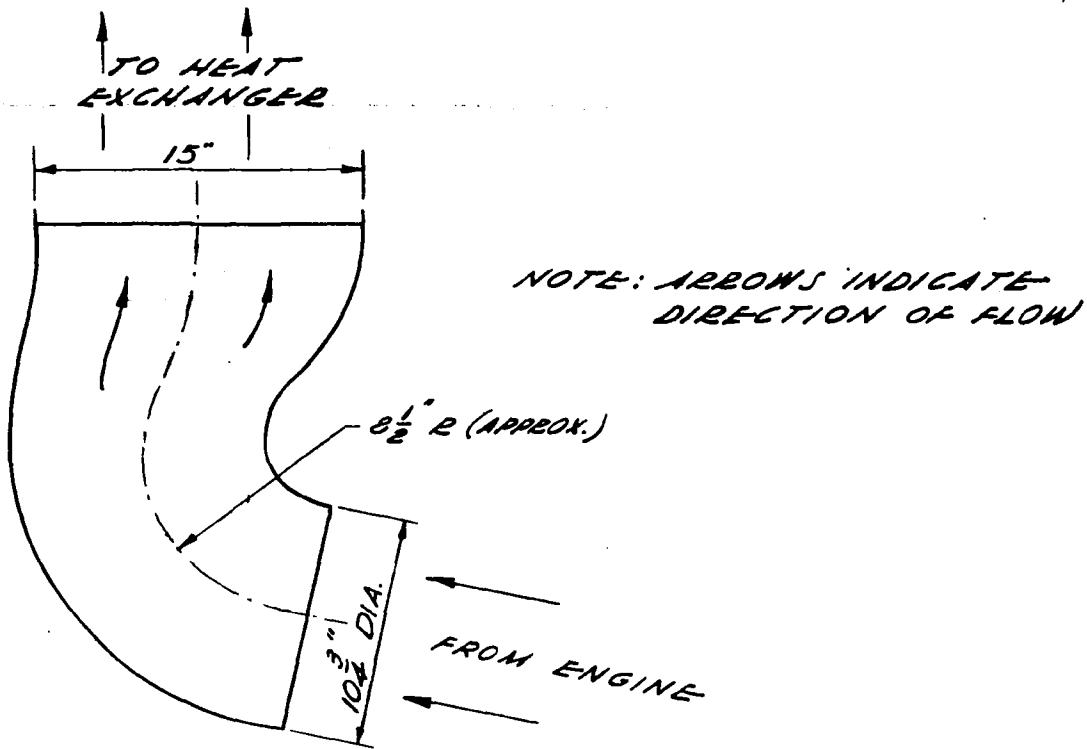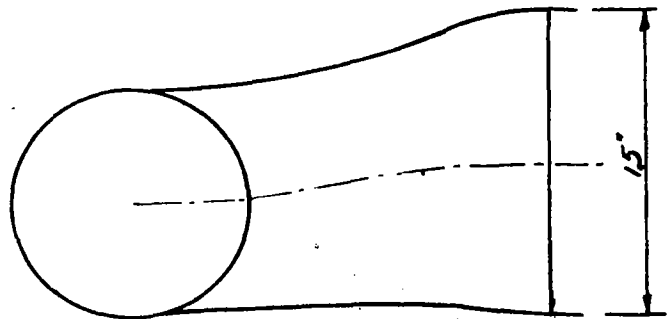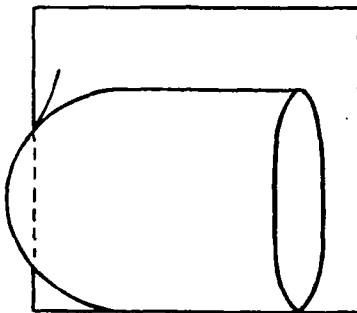
(a) INLET HEADER

FIGURE 4.- HEADERS WHICH DUCT EXHAUST
GAS INTO AND OUT OF THE
HEAT EXCHANGER

FROM HEAT
EXCHANGER

15"

NOTE: ARROWS INDICATE
DIRECTION OF FLOW

6" R (APPROX.)

8" DIA.

8" DIA.

TO TURBO-
SUPERCHARGER
NO. 1

TO TURBO-
SUPERCHARGER
NO. 2

15"
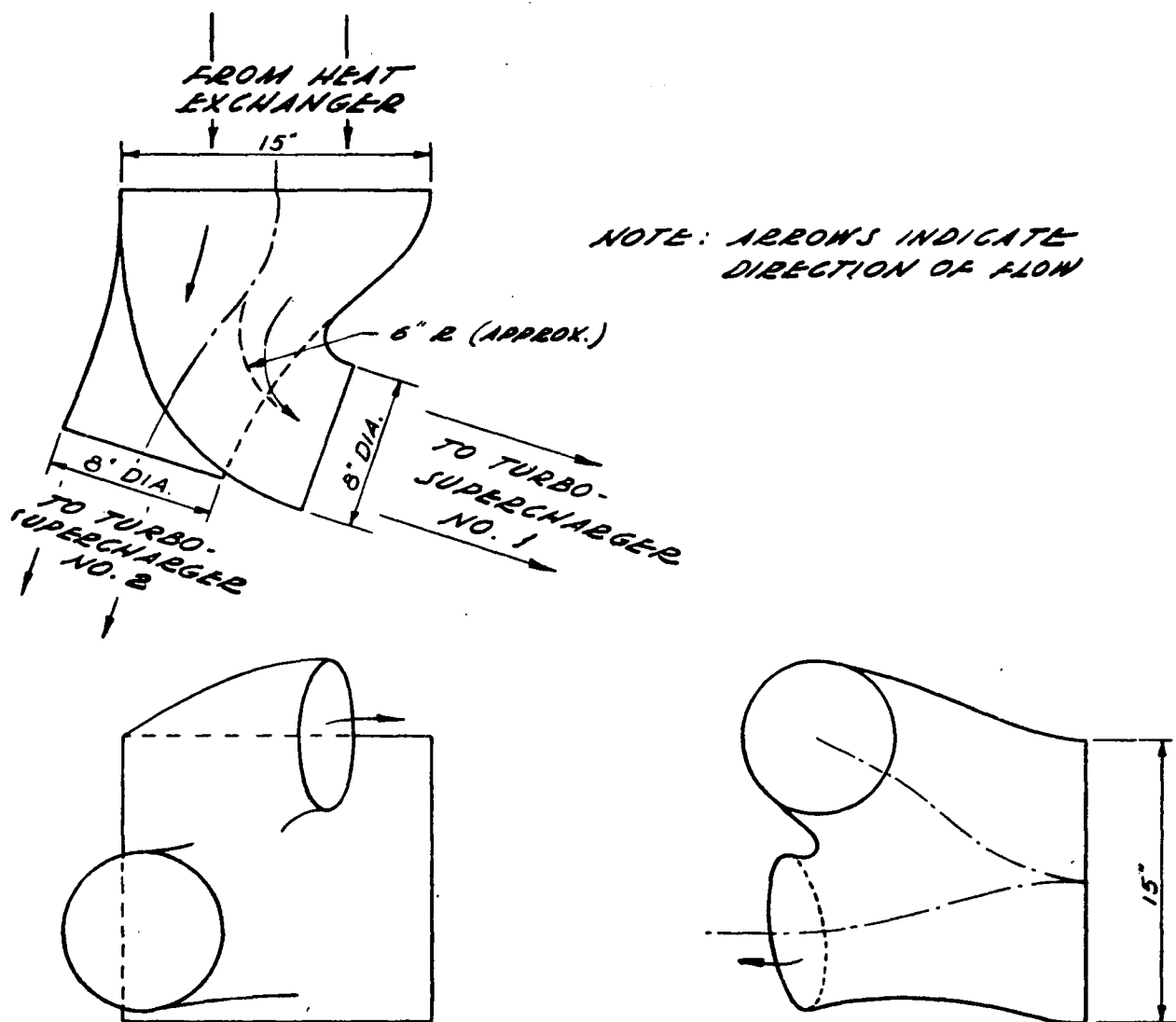
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(b) OUTLET HEADER

FIGURE 4. - CONCLUDED.
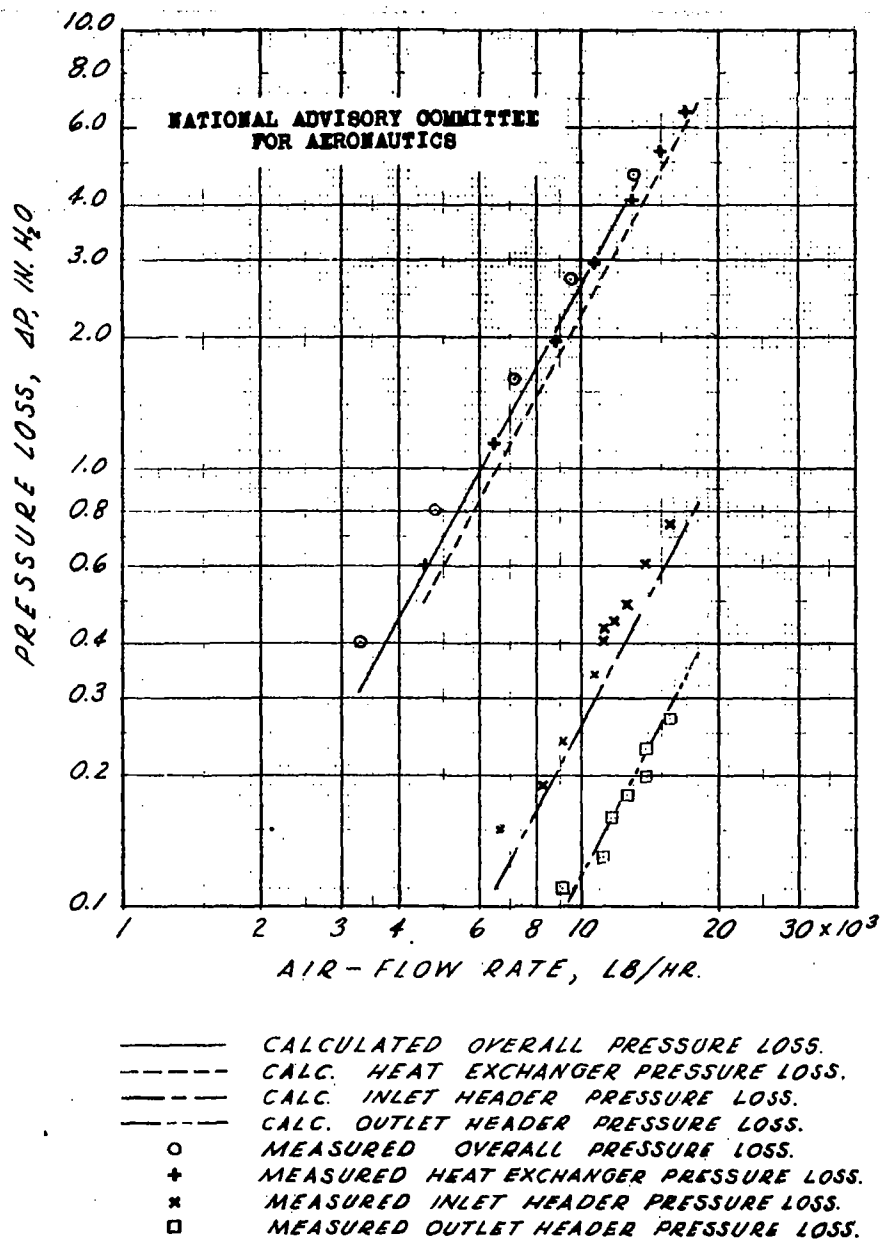
FIGURE 5.- CALCULATED AND MEASURED VALUES OF PRESSURE
        LOSSES IN THE AIR SIDE OF THE HEAT-
        EXCHANGER INSTALLATION AS A FUNCTION OF
        AIR-FLOW RATE.
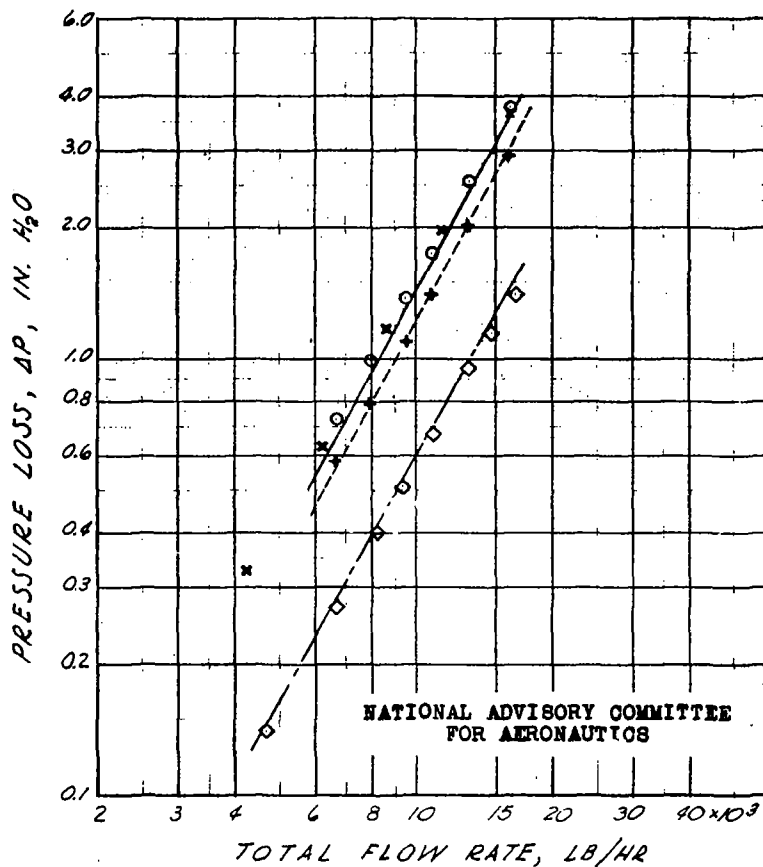
(a) OVERALL AND HEAT-EXCHANGER LOSSES

FIGURE 6 — CALCULATED AND MEASURED VALUES OF
PRESSURE LOSSES IN THE GAS SIDE OF
THE HEAT-EXCHANGER INSTALLATION.
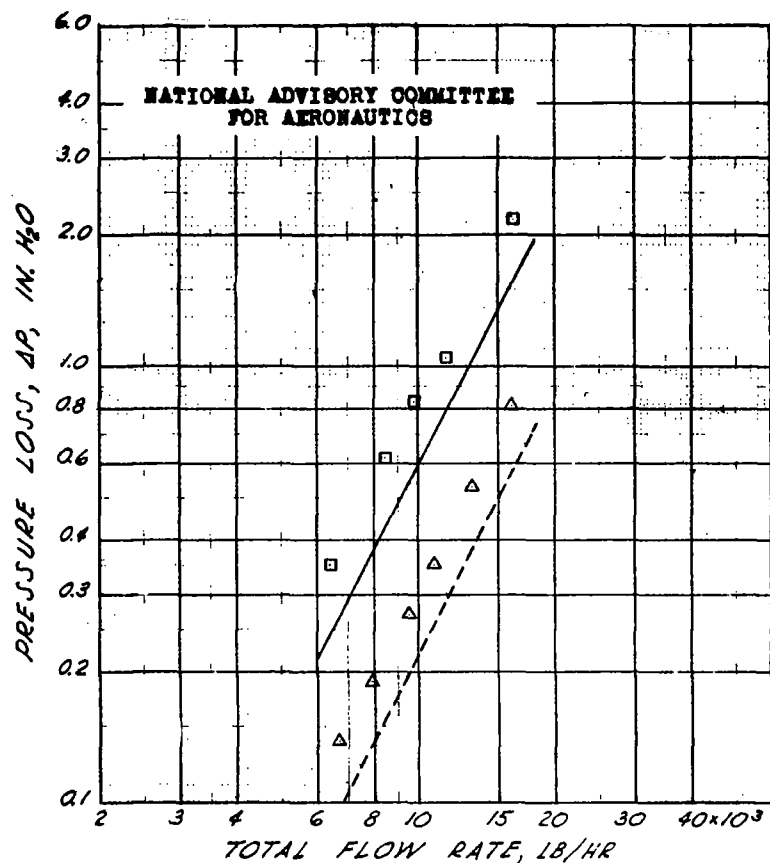
(b) HEADER LOSSES

FIGURE 6. — CONCLUDED.

(a) - AIR SIDE

(b) - GAS SIDE

LEGEND

$\mathcal{L}$ — CENTER LINES OF PRESSURE TUBE TRAVERSES (FOR DETAILS SEE FIG. 11b)

→ DIRECTION OF FLOW
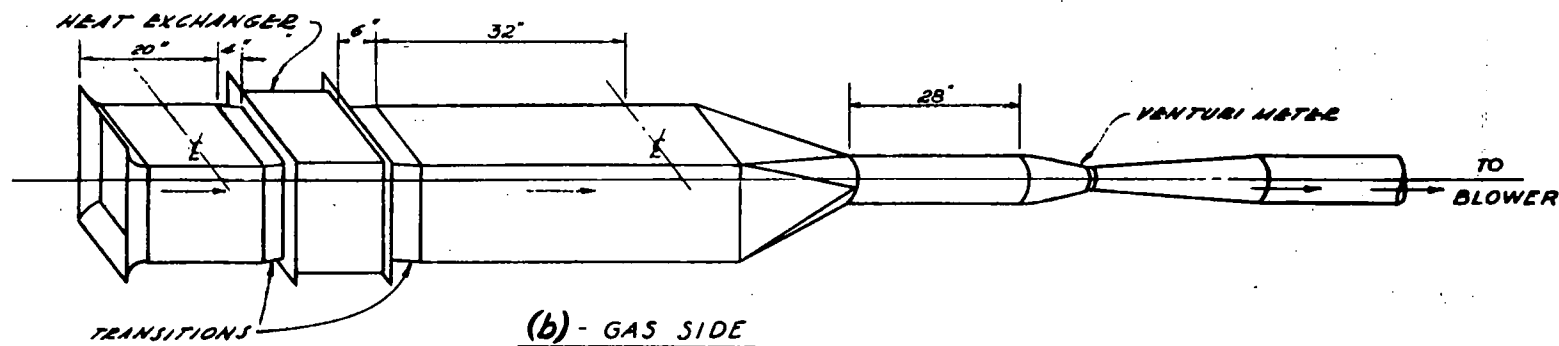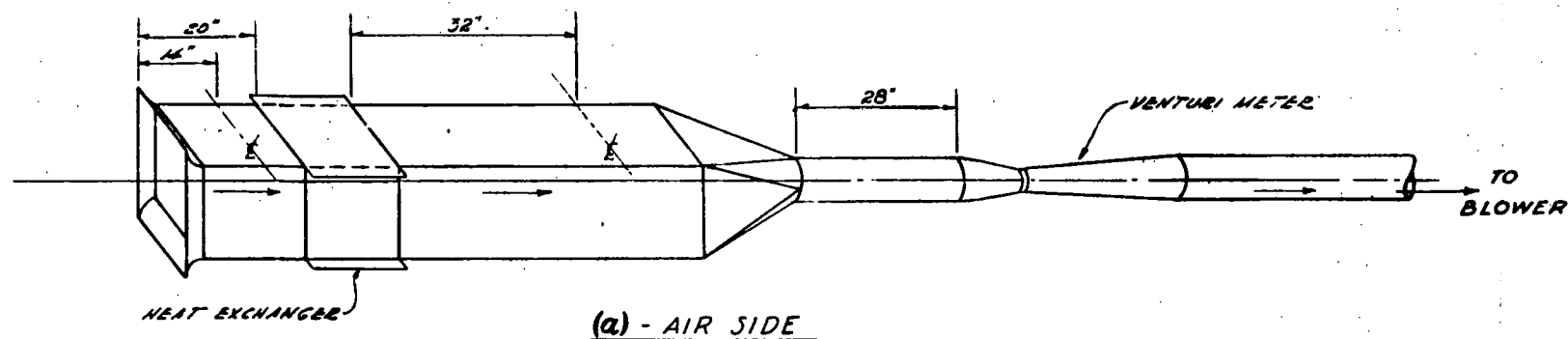
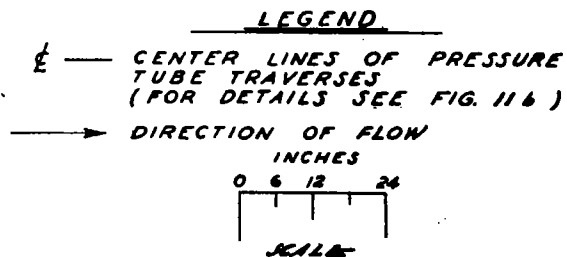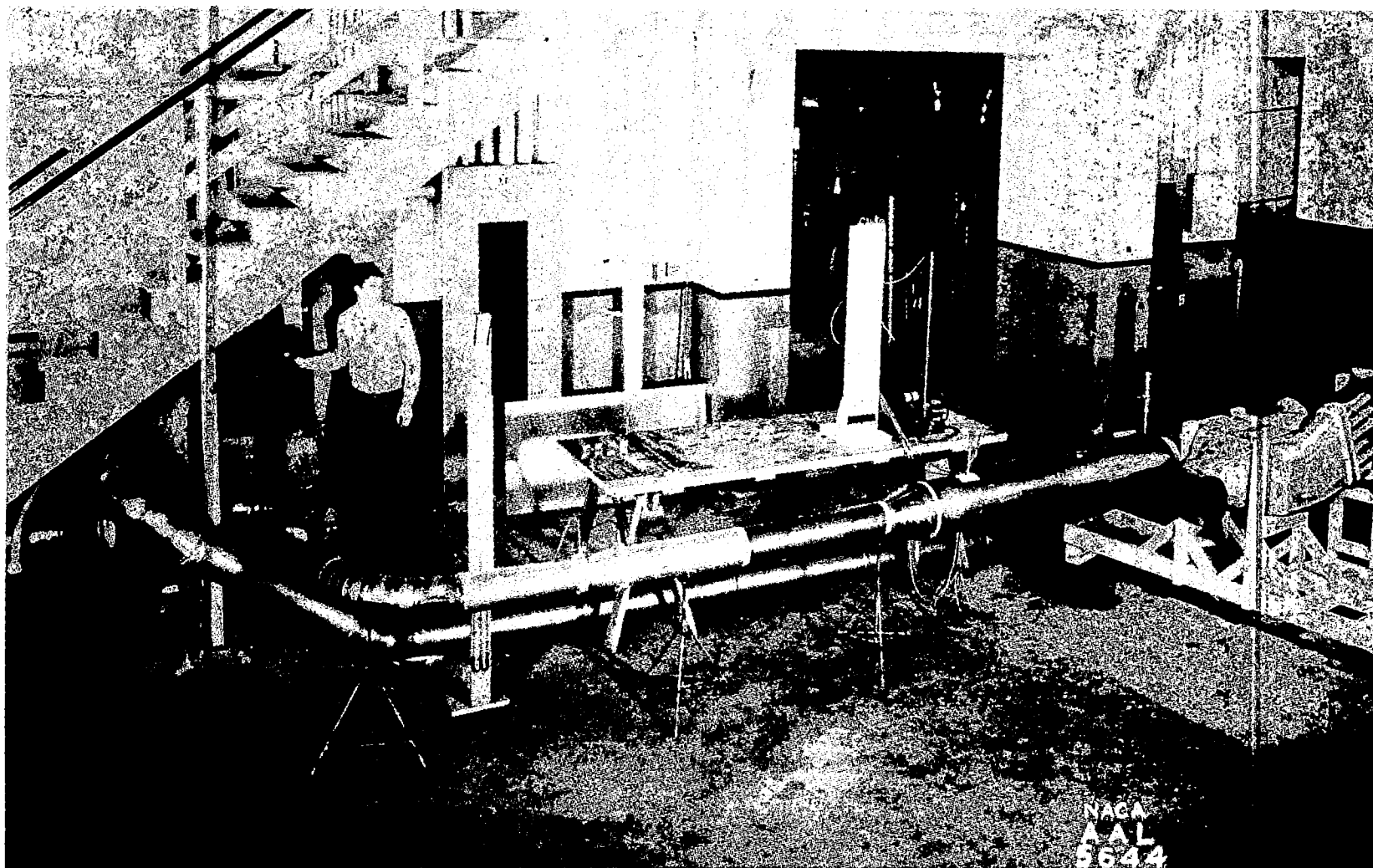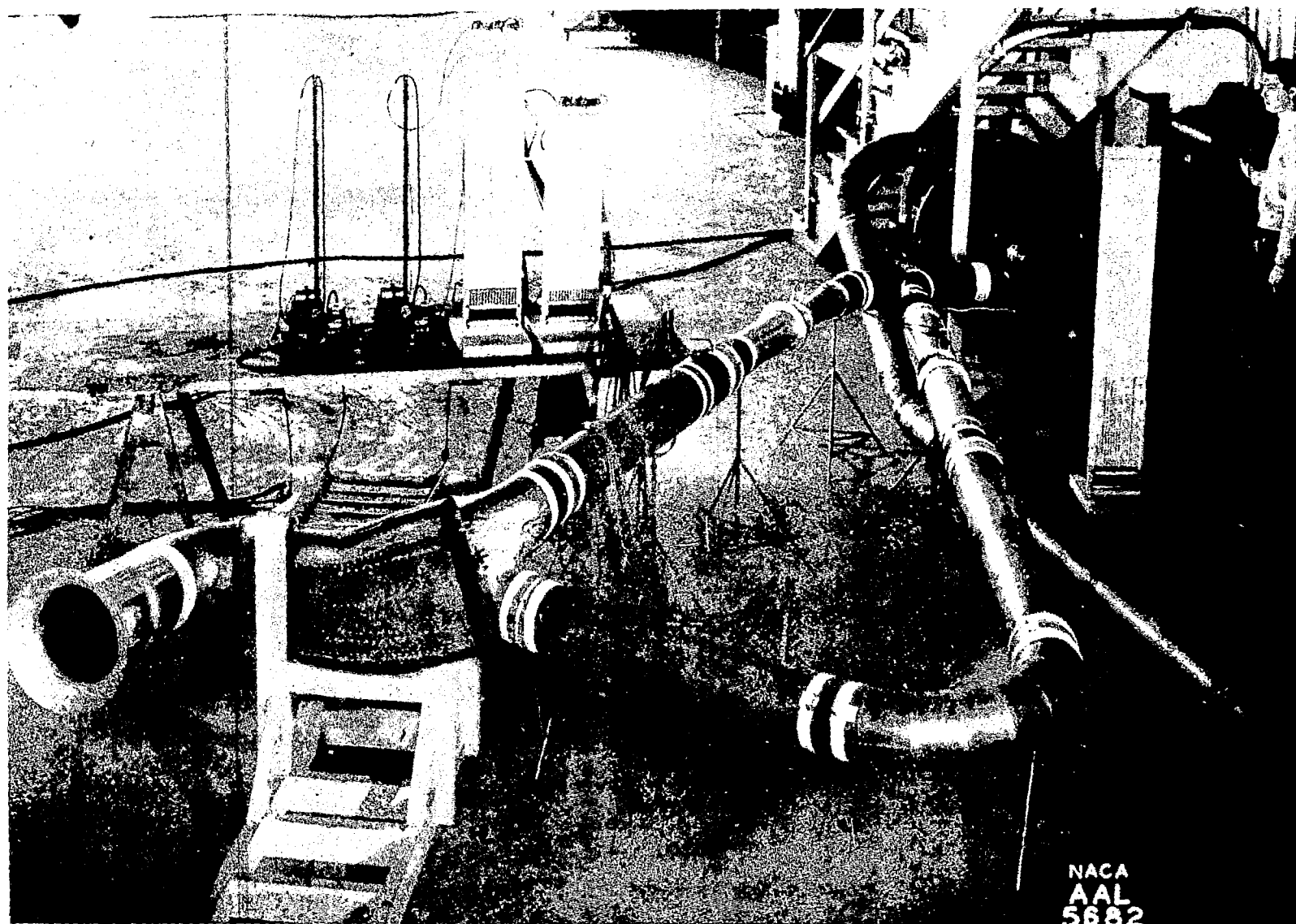NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

FIGURE 7. — LAYOUT OF DUCTING AND LOCATION OF PRESSURE TUBES FOR MEASUREMENT OF PRESSURE LOSS IN ISOLATED HEAT EXCHANGER

Fig. 7

(a) Air side

Figure 8a,b.- Ducting and instrumentation for measuring the over-all pressure loss in the complete heat-exchanger installation.

(b) Gas side   Figure 8.- (Concluded).

Fig. 9

INLET HEADER

SINGLE STATIC
PRESSURE TUBE
AT CENTER OF
DUCT

HEAT EXCHANGER

INLET HEADER

STATIC
PRESSURE TUBE

REMAINDER OF DUCTING
IDENTICAL WITH (a)

OUTLET HEADER

(b) - HEADERS ONLY

OUTLET HEADER

LOCATION OF RAKE FOR
MEASUREMENT OF VELOCITY
DISTRIBUTION. (FOR DETAILS
SEE FIG. 11 a)

PIEZOMETER RINGS

RAKE
(FOR DETAILS
SEE FIG. 11 a)

VENTURI METER

TO
BLOWER

(a) - COMPLETE ASSEMBLY WITH TRAVERSE SECTIONS

LEGEND

₵ — CENTER LINES OF
PRESSURE TUBE TRAVERSES
(FOR DETAILS SEE FIG. 11 b)

← DIRECTION OF FLOW

INCHES
0   6   12   24

SCALE

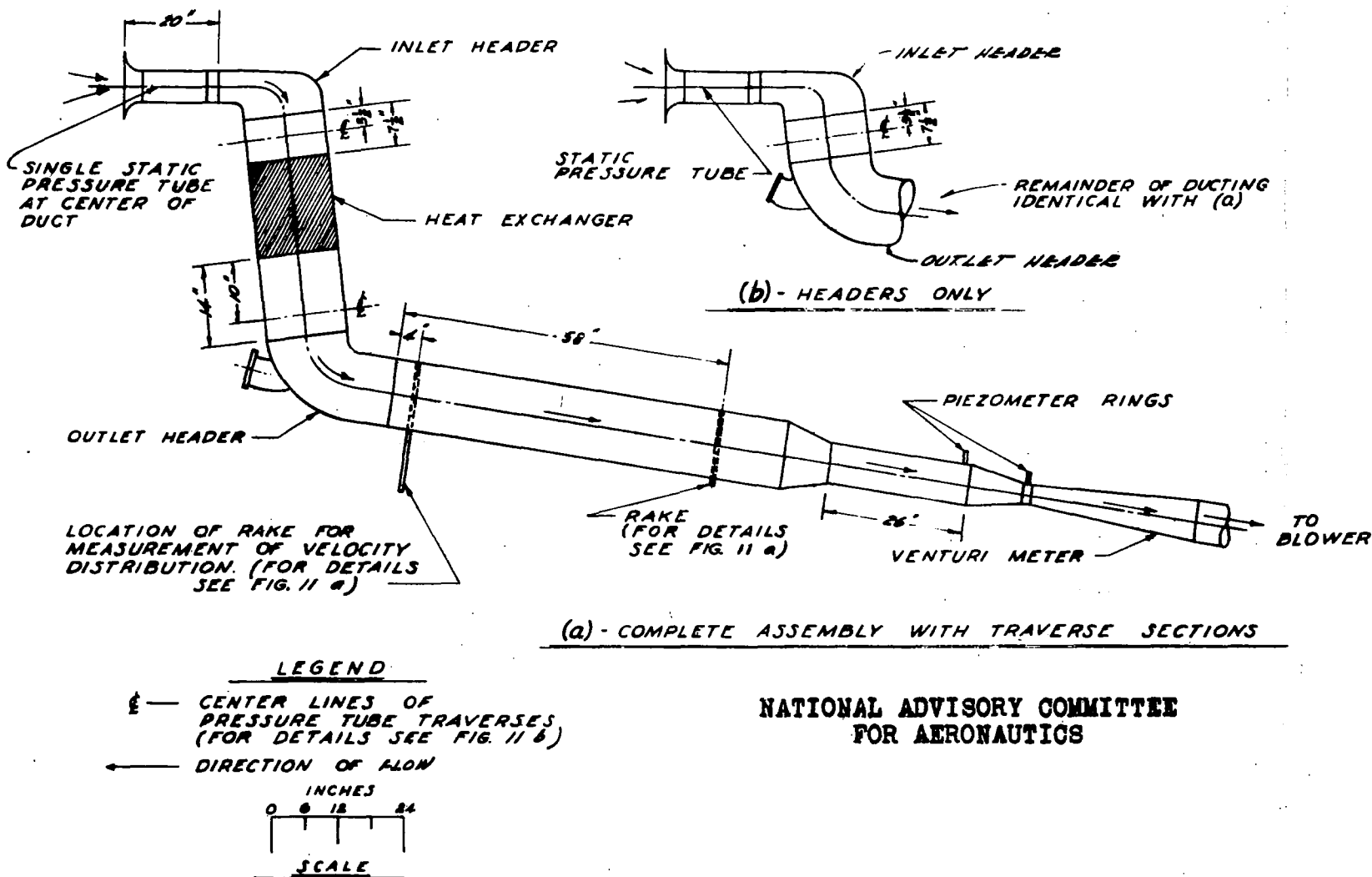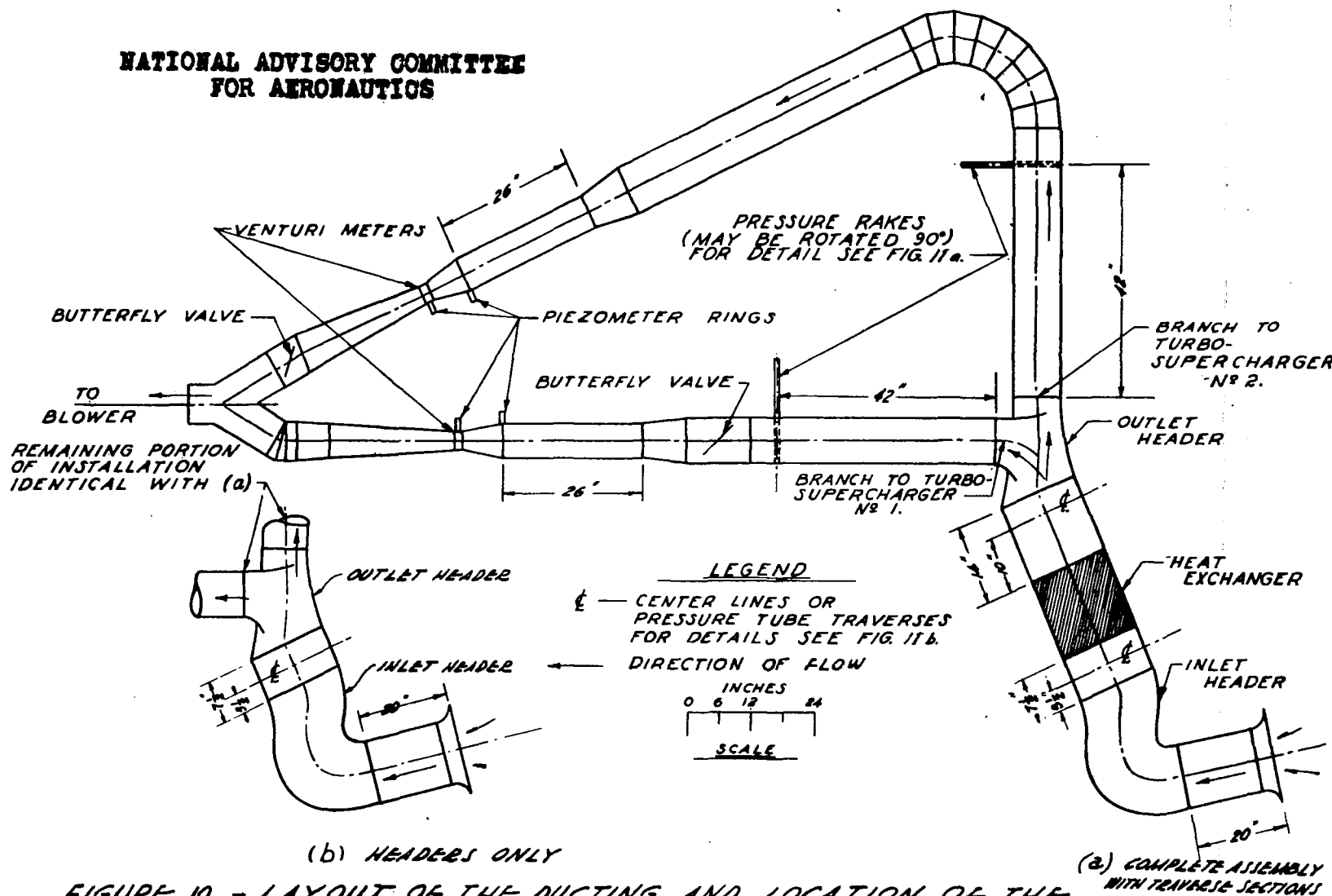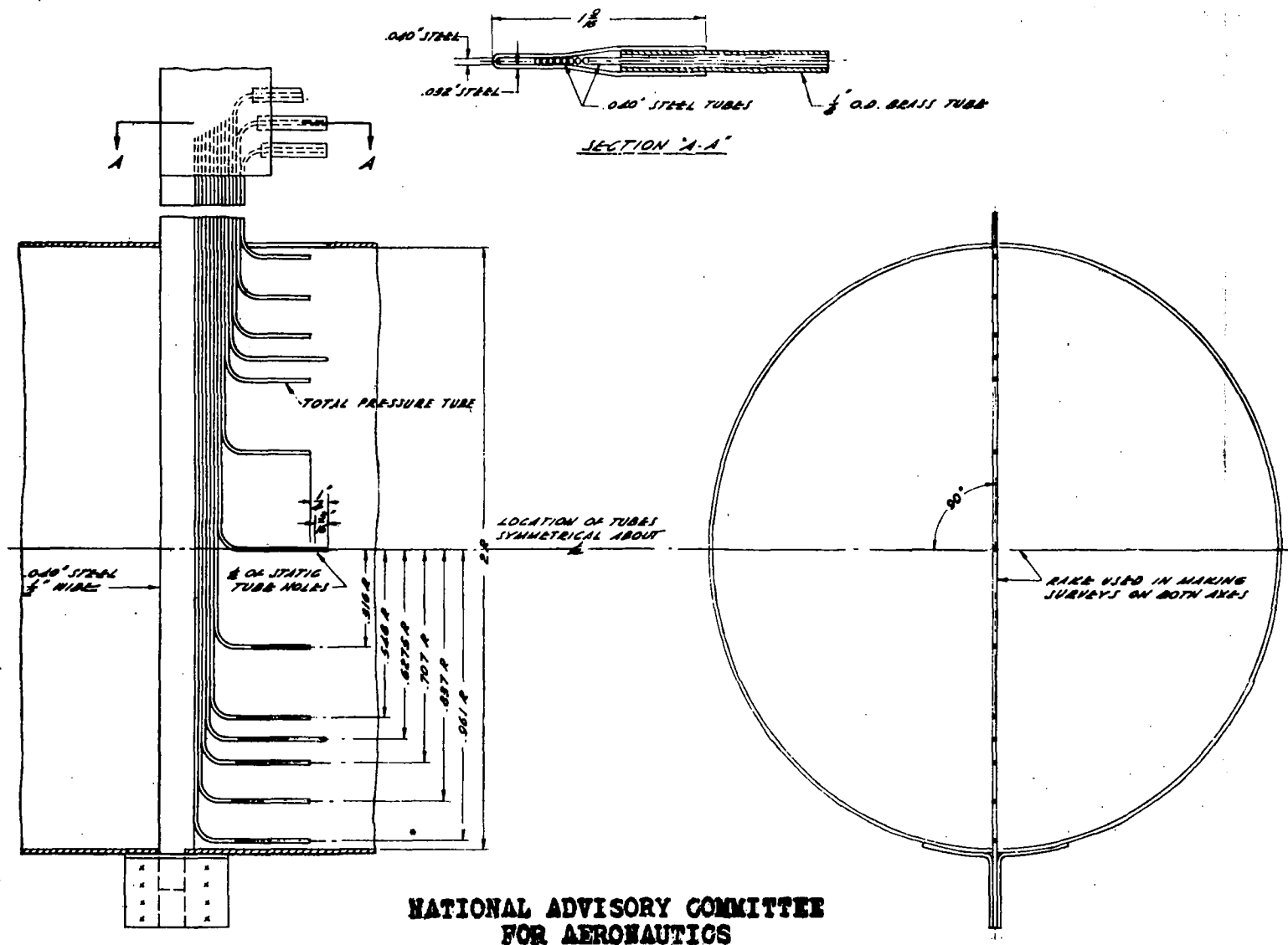NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

FIGURE 9. - LAYOUT OF THE DUCTING AND LOCATION OF THE
INSTRUMENTATION FOR MEASURING THE PRESSURE
LOSSES IN (a) THE PARTS OF THE AIR SIDE OF THE COMPLETE
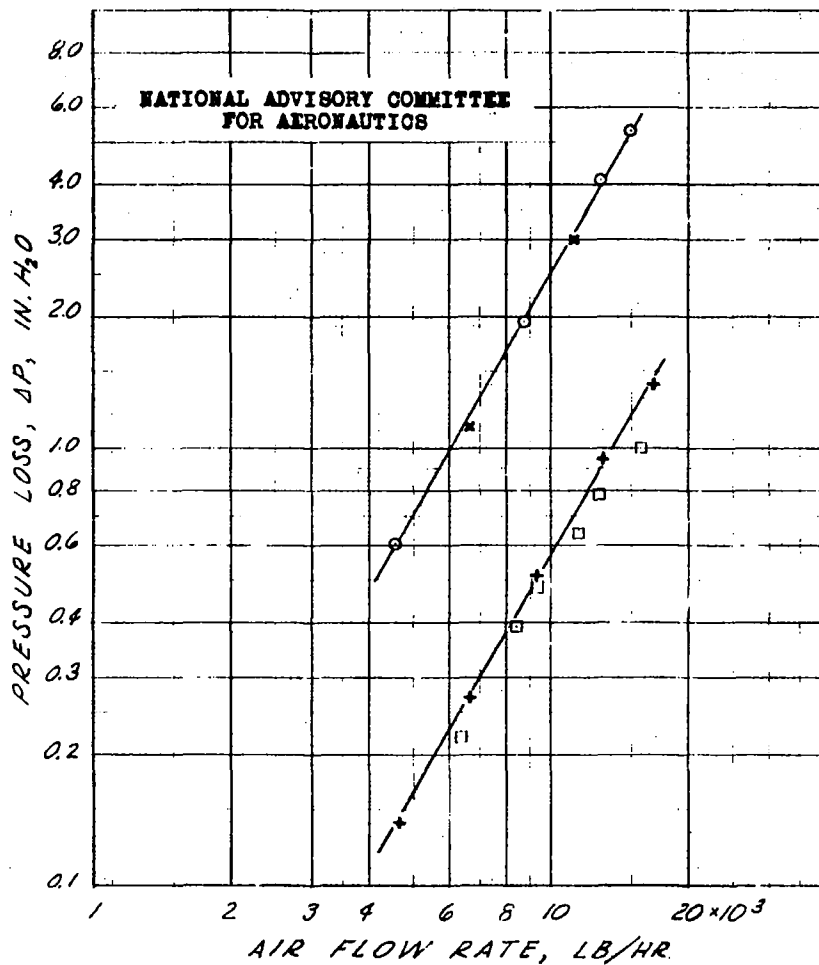HEAT-EXCHANGER ASSEMBLY AND (b) THE AIR HEADERS.

VENTURI METERS

PRESSURE RAKES
(MAY BE ROTATED 90°)
FOR DETAIL SEE FIG. 11a.

BUTTERFLY VALVE

PIEZOMETER RINGS

26'

BRANCH TO
TURBO-
SUPERCHARGER
N° 2.

TO
BLOWER

BUTTERFLY VALVE

42"

OUTLET
HEADER

REMAINING PORTION
OF INSTALLATION
IDENTICAL WITH (a)

26'

BRANCH TO TURBO-
SUPERCHARGER
N° 1.

OUTLET HEADER

HEAT
EXCHANGER

LEGEND

¢ — CENTER LINES OR
PRESSURE TUBE TRAVERSES
FOR DETAILS SEE FIG. 11 b.

INLET HEADER

DIRECTION OF FLOW

INLET
HEADER

INCHES
0 6 12   24

SCALE

20'

(b) HEADERS ONLY

(a) COMPLETE ASSEMBLY
WITH TRAVERSE SECTIONS

FIGURE 10. — LAYOUT OF THE DUCTING AND LOCATION OF THE
INSTRUMENTATION FOR MEASURING THE PRESSURE
LOSSES IN (a) THE PARTS OF THE GAS SIDE OF THE
COMPLETE HEAT-EXCHANGER ASSEMBLY AND (b) THE GAS HEADERS.
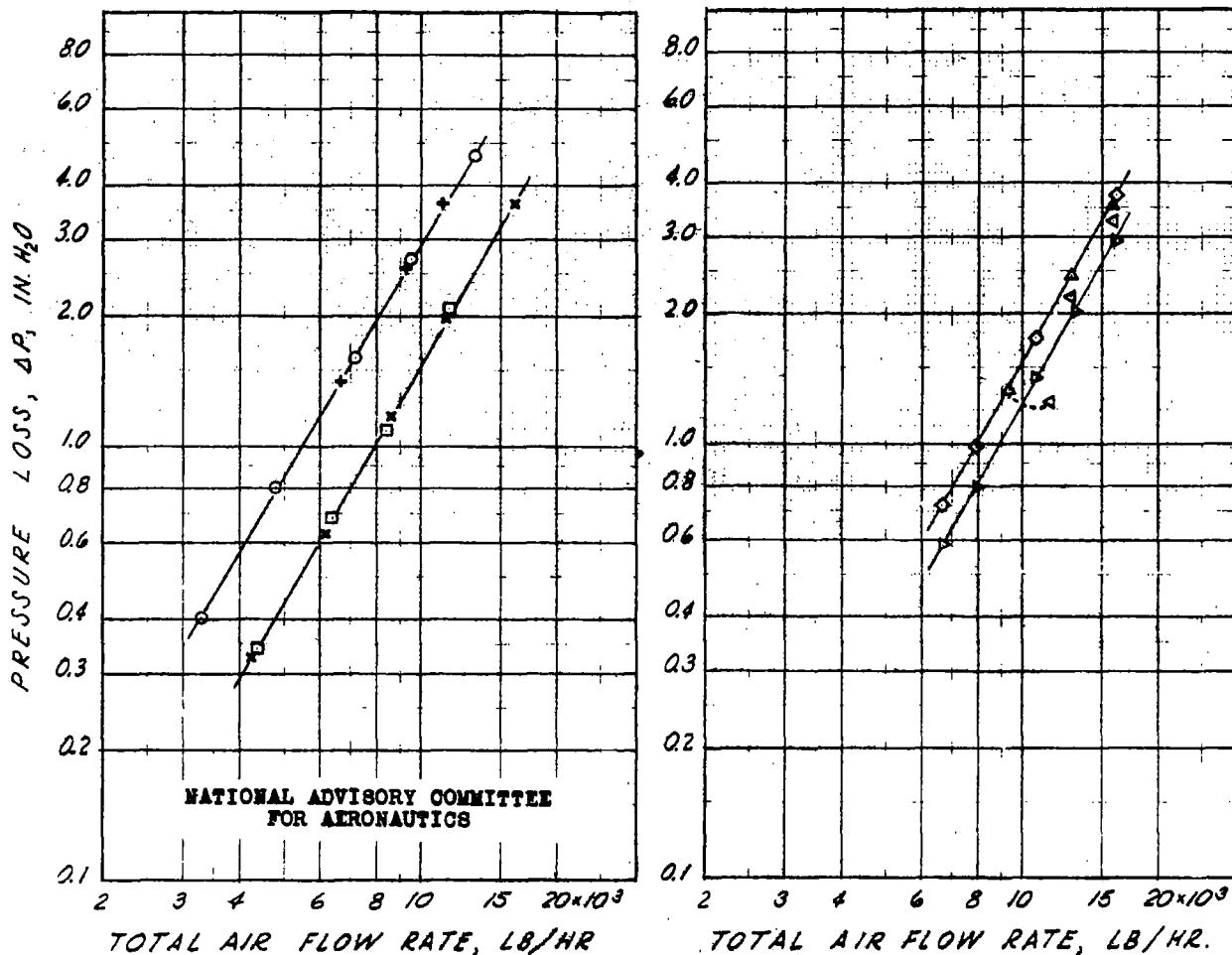
Fig. 10

SECTION "A-A"

TOTAL PRESSURE TUBE

LOCATION OF TUBES
SYMMETRICAL ABOUT

.040" STEEL
⅛" WIDE

℄ OF STATIC
TUBE HOLES

RAKE USED IN MAKING
SURVEYS ON BOTH AXES

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(a) PRESSURE RAKES

FIGURE II. — INSTRUMENTATION DETAILS

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MOTION

MOTION

TUBES SHOWN IN DETAILS 1
AND 2 ARE INTERCHANGEABLE

0.050" D.

⅛"D.

SILVER SOLDER

0.020"

SECTION "A-A"

DETAIL 1
SHIELDED TOTAL PRESSURE TUBE

⅛"D.

DETAIL 2
STATIC PRESSURE TUBE

(b) TRAVERSABLE SHIELDED TOTAL AND STATIC PRESSURE TUBES

FIGURE 11.- CONCLUDED

MEASURED PRESSURE LOSS IN HEAT
EXCHANGER WITH STRAIGHT HEADERS (SEE FIG. 7)
  O   AIR SIDE
  +   GAS SIDE
MEASURED PRESSURE LOSS IN HEAT EXCHANGER
WITH INSTALLATION HEADERS (SEE FIGS 9a & 10a)
  ✗   AIR SIDE
  ☐   GAS SIDE

FIGURE 12.- COMPARISON OF MEASURED VALUES OF PRESSURE
LOSS IN THE HEAT EXCHANGER WITH AND
WITHOUT HEADERS AS A FUNCTION OF FLOW RATE.

FIGURE 13.— COMPARISON OF MEASURED VALUES OF OVERALL PRESSURE
LOSS THROUGH THE HEAT-EXCHANGER INSTALLATION WITH
AND WITHOUT TRAVERSING SECTIONS BETWEEN HEADERS AND
HEAT EXCHANGER.

FIGURE 14.- COMPARISON OF CALCULATED AND MEASURED
        VALUES OF ΔP/q₀ FOR THE AIR-SIDE PARTS
        OF THE HEAT-EXCHANGER INSTALLATION.

Fig. 15

$\Delta P/q_0$

9%

-5%

-30%

30%

CALCULATED

CALCULATED

4%

0

INLET   EXCHANGER   OUTLET   TURBO 1   TURBO 2
                       ELBOW   O V E R A L L
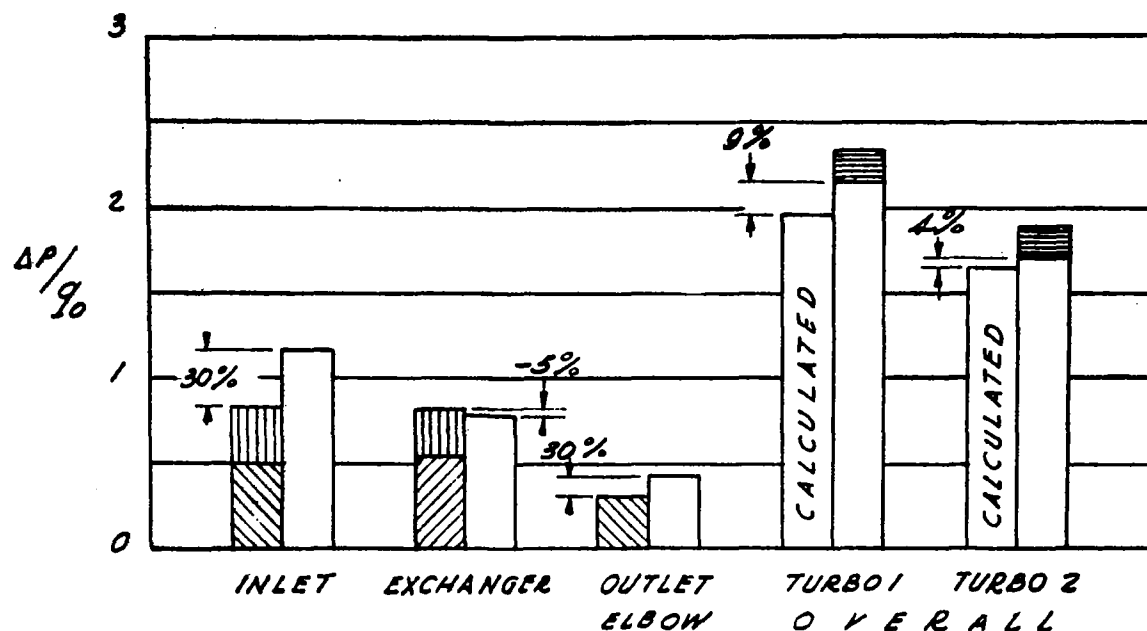
CALCULATED FRICTION LOSS.

CALCULATED ELBOW LOSS.

CALCULATED EXPANSION LOSS.

EXPERIMENTAL LOSS.

EXPERIMENTAL ERROR IN
MEASURING COMPONENT LOSSES

$$\% = \frac{EXPERIMENTAL - CALCULATED}{EXPERIMENTAL}$$

INLET VELOCITY HEAD, q_0, = 1.03 IN. H_2O
AT FLOW RATE OF 12,000 LB. PER HR.

FIGURE 15.- COMPARISON OF CALCULATED AND MEASURED VALUES OF
$\Delta P/q_0$ FOR THE EXHAUST-GAS-SIDE PARTS OF THE
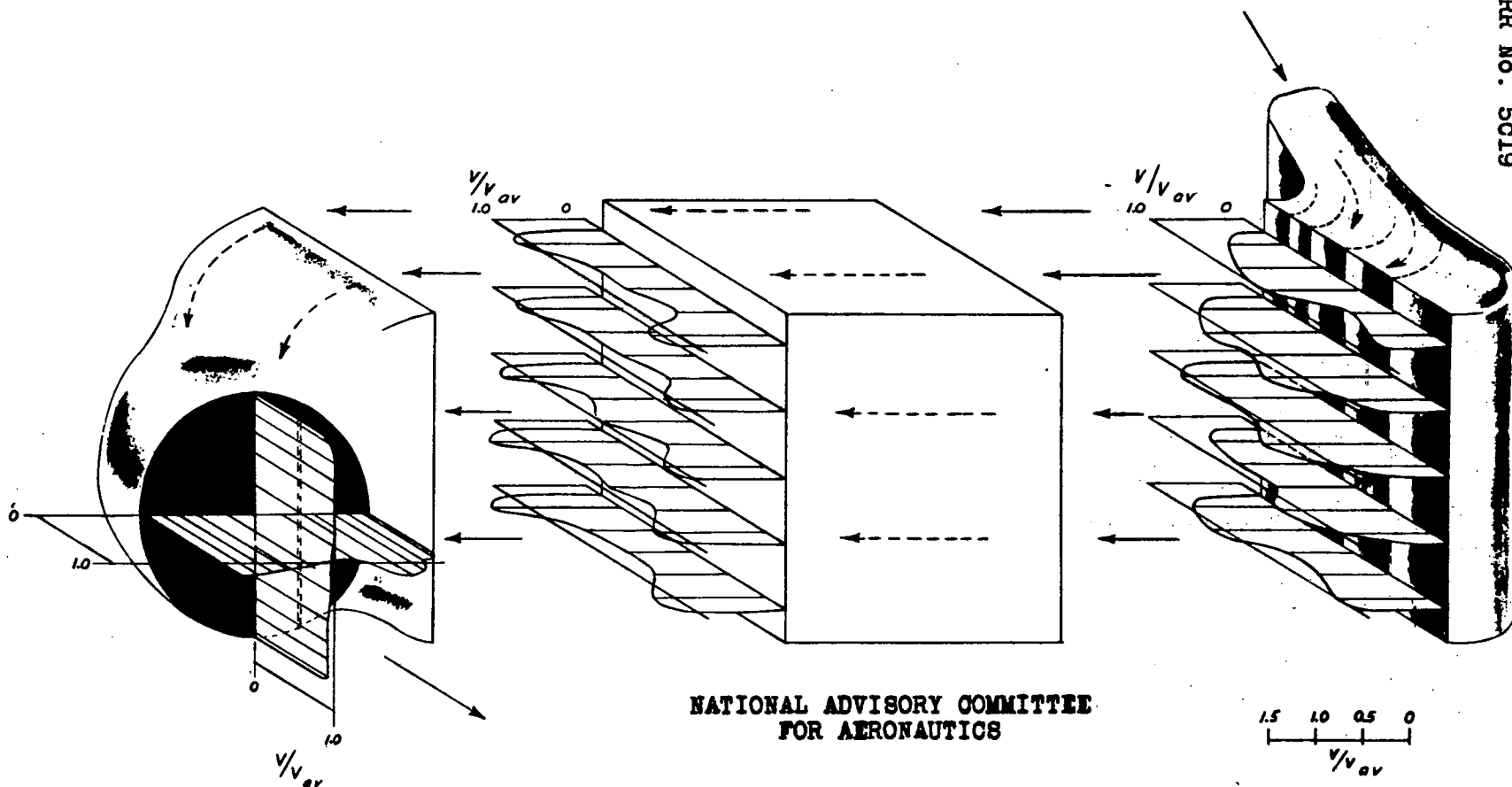HEAT-EXCHANGER INSTALLATION.

FIGURE 16.- VELOCITY DISTRIBUTIONS IN THE AIR SIDE OF THE COMPLETE
HEAT-EXCHANGER INSTALLATION.

Fig. 16

Fig. 17
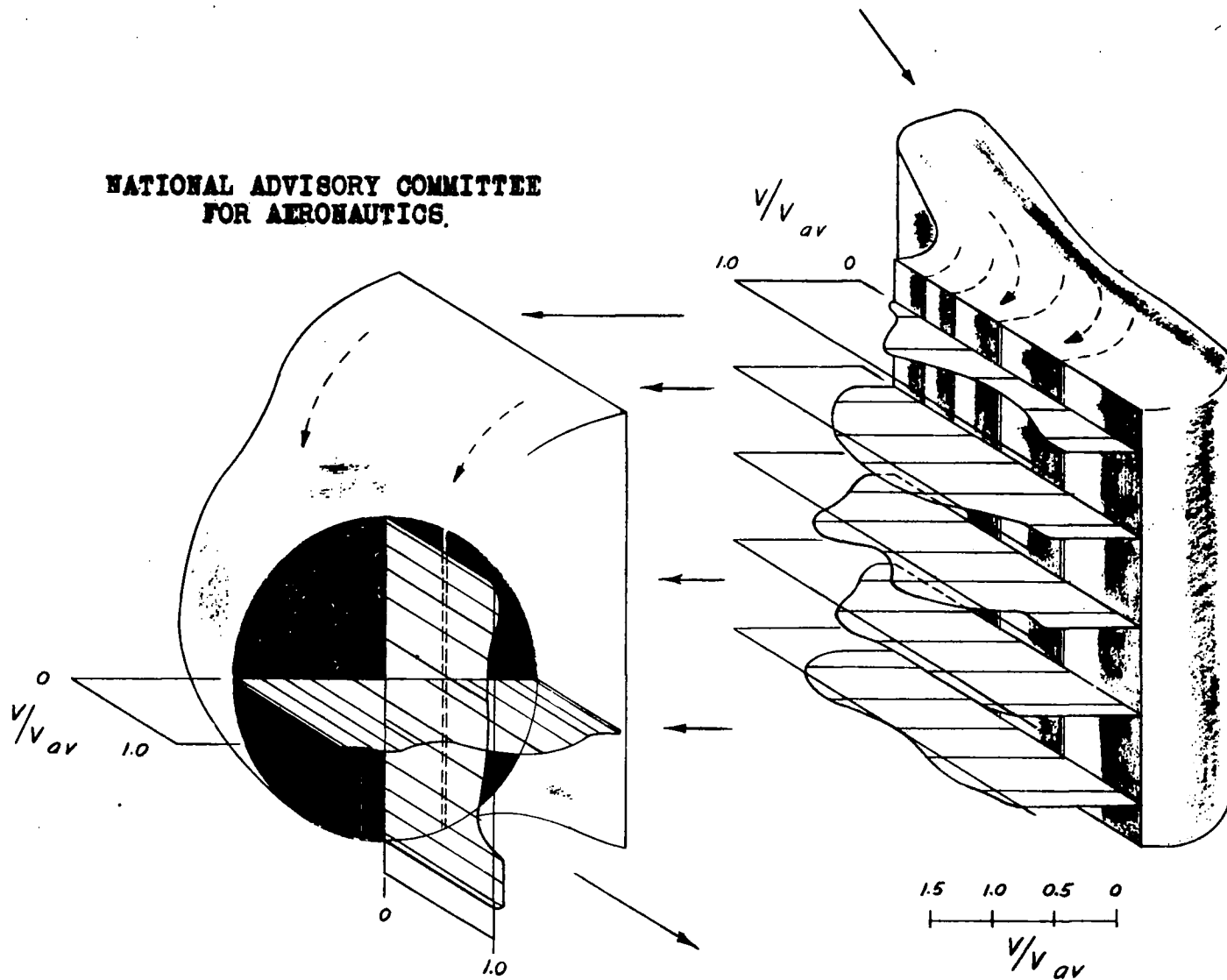
NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS.

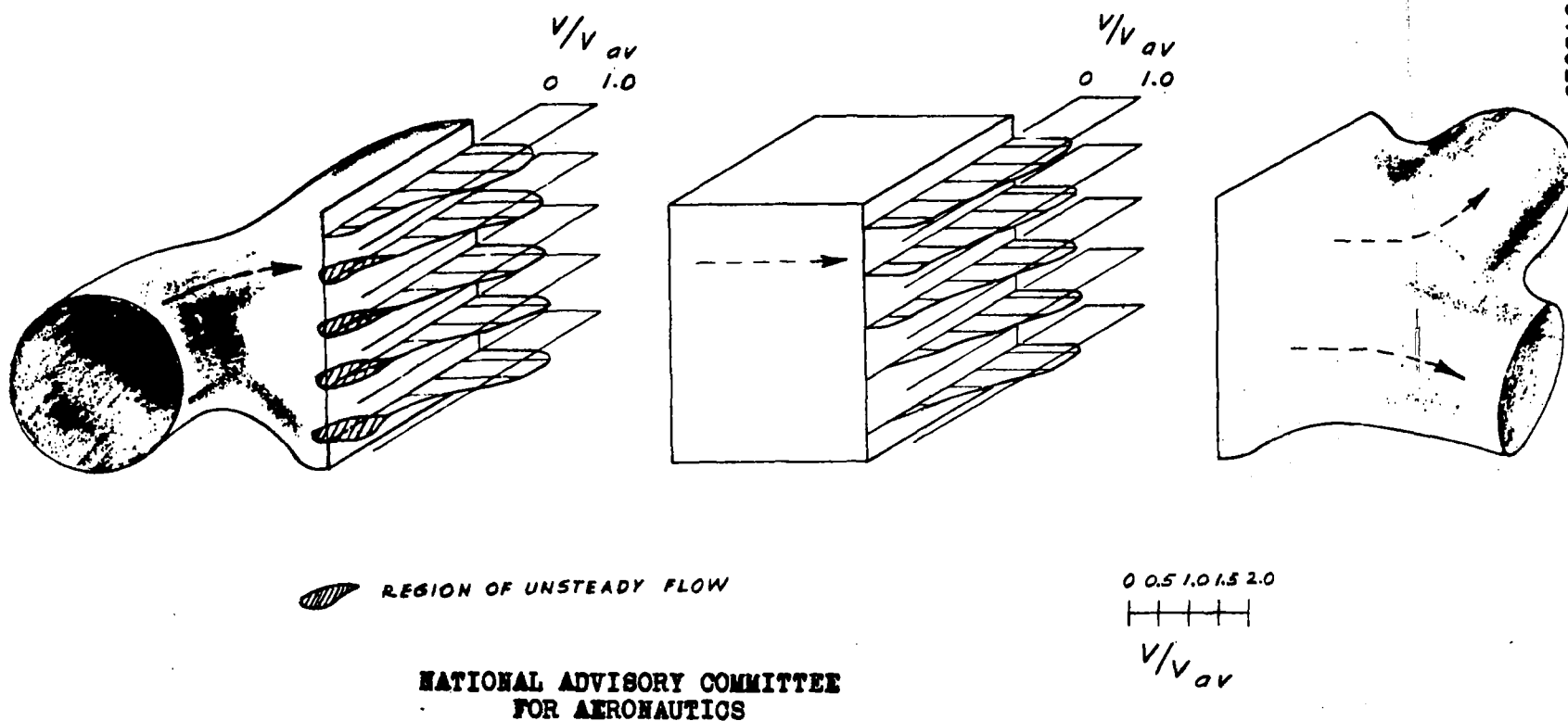FIGURE 17.- VELOCITY DISTRIBUTIONS IN THE AIR HEADERS WITH
THE HEAT EXCHANGER REMOVED

REGION OF UNSTEADY FLOW

FIGURE 18.- VELOCITY DISTRIBUTIONS IN THE GAS SIDE OF
THE COMPLETE HEAT-EXCHANGER INSTALLATION.

Fig. 18

V/V_{av}

0   1.0   2.0

REGION OF UNSTEADY FLOW

0 0.5 1.0 1.5 2.0

V/V_{av}
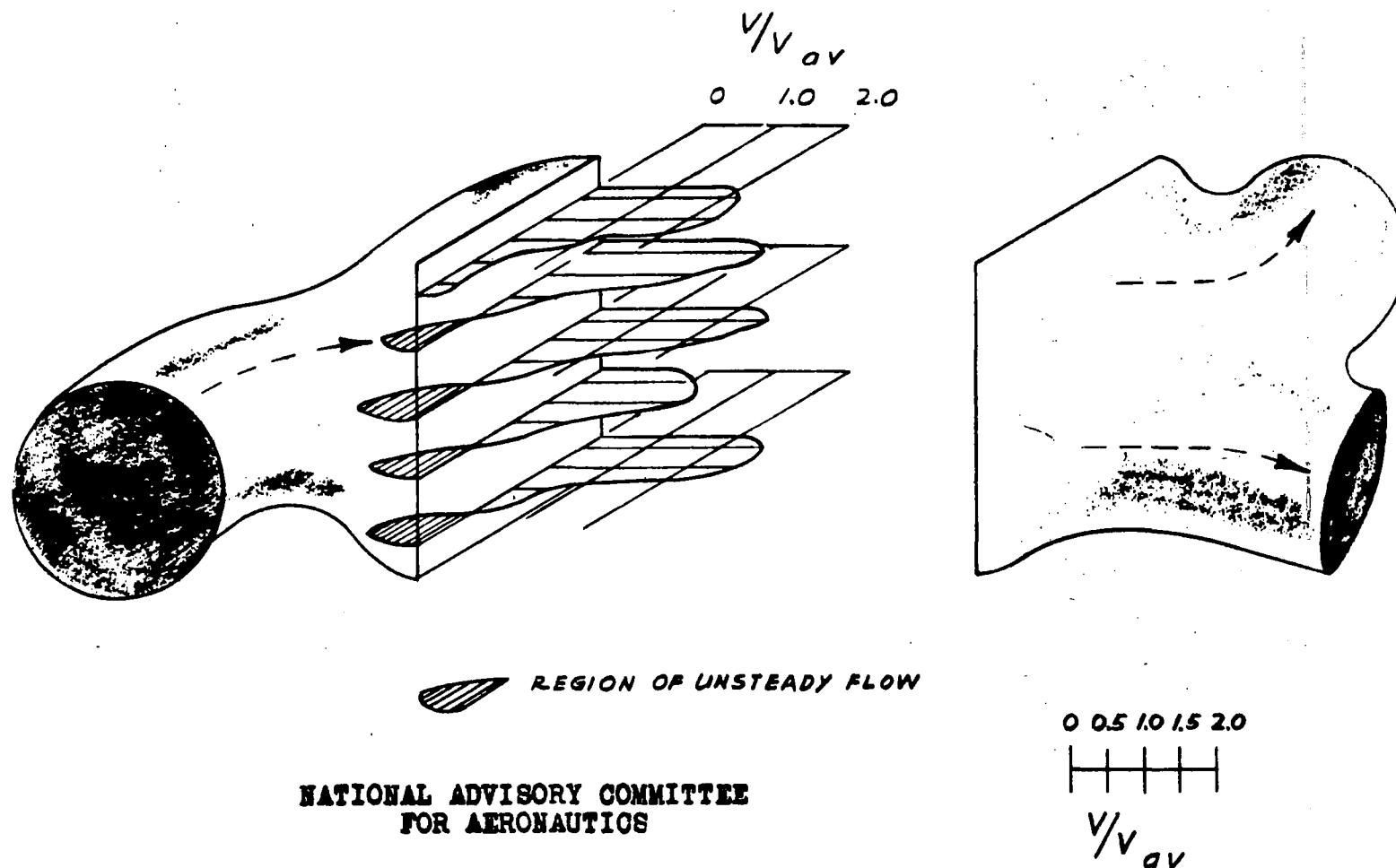
FIGURE 19.- VELOCITY DISTRIBUTIONS IN THE GAS INLET
HEADER WITH THE HEAT EXCHANGER REMOVED.

Fig. 19

DIVISION: Power Plants, Reciprocating (6)
SECTION: Cooling (1)
CROSS REFERENCES: Heat exchangers (48400); Diffusers (30200); Flow through ducts (41200)

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ABSTRACT

Tests and calculations were conducted for obtaining data useful in the design of efficient ducting systems. Analytical values were determined by adding together losses calculated from data on elbows, diffusers, and straight ducts. For ducts similar to tested cross-flow plate-type heat exchanger, calculated losses will agree with experimental values. Losses in poorly designed elbows can be predicted, but experimental values of losses are subject to the uncertainties of pressure measurement in regions of uneven flow distribution.

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